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VIBRATION TESTS ON HARBOUR PERSONNEL BOAT.(U)

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Structures Technical Memorandum 274

VIBRATION TESTS ON HARBOUR PERSONNEL BOAT

G. LONG AND C.M. BAILEY.

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Summary

Excessive propeller shaft bearing wear has occurred on a series of 10 metre harbour personnel boats constructed for the Royal Australian Navy.

An investigation of the problem revealed that the wear was caused by a high-order whirling mode of the shaft excited by propeller-hull interaction.

It was not practicable to alter this whirling vibration significantly, but the wear rate was reduced to an acceptable level by changing the bearing materials.

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16. ABSTRACT:

Excessive propeller shaft bearing wear has occurred on a series of 10 metre harbour personnel boats constructed for the R.A.N.

An investigation of the problem revealed that the wear was caused by a high-order whirling mode of the shaft excited by propeller-hull interaction.

It was not practicable to alter this whirling vibration significantly but the wear rate was reduced to an acceptable level by changing the bearing materials.

CONTENTS

1. INTRODUCTION.

The 10 metre harbour personnel boats were designed by the Navy for general use in harbour and also to be carried on board ships. For the latter use, it is normal practice to test the engine and transmission frequently, and this can involve operating the propeller shafts in dry un-lubricated bearings. Mainly for this reason, Tufnol was chosen for the bearing material, and, since this material swells in water, the bearing clearances were larger than normal. The shaft diameter is 42.9 mm (1.688") and the material used is aluminium bronze.

The initial design of the boat used two propellers of 508 mm (20 in.) diameter driven by shafts each supported by two bearings. Subsequently, intermediate bearings were fitted as indicated in FIG. 1. For various reasons, including that of operation in shallow water the propeller tip clearance was chosen to be the minimum practical value.

During acceptance trials of the boats it was found that the aft bearings experienced excessive wear, and this was attributed to the high level of vibration experienced on these craft. These laboratories were asked to investigate this problem and to recommend a solution.

2. DETAILS OF INVESTIGATION.

Initial tests were made on a boat at sea by recording the acceleration levels above the P-bracket attachment to the hull. No strong indication of resonant vibrations was detected during these tests, but detailed analysis of the frequency content of the signals indicated significant vibration response at shaft frequency and blade frequency.

Vibration tests were conducted on the propeller shaft system, when the boat was out of the water, to determine the natural frequencies and mode shapes of vibration. The three lowest frequency modes measured are illustrated in FIG. 1.

The transverse mode of vibration at 25.7 Hz. occurs at a frequency just greater than the maximum shaft rotational frequency, hence an initial modification was made to stiffen the P-bracket in the lateral direction. This was achieved by bolting an additional transverse bracket between the bearing housing and the bottom of the boat close to the centreline. Subsequent tests indicated that this modification had negligible effect on the bearing wear.

An accelerometer was attached to the P-bracket close to the bearing housing and oriented so that its sensitive axis was in the transverse direction. Sea trials were conducted both before and after fitting the stiffening bracket, and the resulting vibration data were analysed on a real-time spectrum analyser (FIG. 2A & 2B). Graphs of the response at shaft frequency and blade frequency are reproduced in FIG. 3 & 4. In these figures there is no indication of any resonant response at shaft frequency, but there is an apparent heavily damped resonance at approximately 69 Hz. (61 Hz. in FIG. 3), excited by blade frequency forces. These frequencies are sufficiently close to the vertical vibration mode at 67 Hz. (see FIG. 1) to suspect a higher order whirling mode of vibration, in which the shaft bending mode is excited by forces induced on the propeller blades as they pass close to the hull. The mode is not well excited by this action and appears to be heavily damped.

Such a mode of vibration has been described in reference 1. In this reference the transverse vibration of much larger propeller shafts is discussed, and it is pointed out that these vibrations are often caused by blade frequency forces exciting higher order modes of the shafting. Often this problem is aggravated by excessive bearing clearance and bearing misalignment, which create longer lengths of unsupported shafting than was intended. Also described in this reference is the heavily damped nature of the vibration response.

At this stage it was concluded that the bearing wear was being caused by transverse vibration of the shaft in a mode similar to that indicated in FIG. 1 at 67 Hz., and being excited by forces on the propeller blades as they pass close to the hull. The problem was therefore to find an economical solution.

Although the additional stiffening of the P-bracket reduced the vibration at shaft frequency, the level of higher frequency vibration was increased, as is evident in FIG. 2B, and the bearing wear rate was unaffected. Consequently the stiffening was removed for subsequent tests. One P-bracket however was stiffened by a carbon fibre streamlined sleeve which effectively increased the transverse frequency of vibration to approximately 33 Hz. This modification was primarily intended to reduce the possibility of a transverse resonance occurring in the P-bracket and was not seen as a possible cure of the bearing wear problem. Carbon fibre was chosen as a stiffening material because of its ease of application and also to investigate its durability in a severe marine environment. This aspect of the work is described in detail in reference 2.

3. METHOD OF SOLUTION.

Three possible methods of reducing the wear rate of the bearings were considered. These were:

1. Reduce the vibration by reducing the excitation force.

2. Avoid resonance by changing the natural frequency of the shafting system to move the resonance out of the operating range.

3. Reduce the bearing clearances and possibly change the bearing materials.

These are discussed separately below.

3.1 Reduction of excitation force.

As previously noted it was concluded that the input excitation force was caused by the propeller blades passing close to the hull. This force can only be reduced by increasing the propeller tip clearance, either by changing the angle of the shaft or reducing the propeller diameter. Neither of these solutions was considered practicable because of the cost involved in modifying all the boats.

3.2 Avoidance of resonance by changing the shaft natural frequency.

The simplest method of increasing the shaft transverse bending frequency is to substitute a steel shaft for the bronze. Since the densities of the materials are comparable, and the steel has a Young's modulus approximately twice that of the bronze, the natural frequency of the steel shaft would be expected to increase significantly. This approach avoided any modification to the boat itself. The natural frequencies of the original bronze shaft and the replacement steel shaft were measured and are compared in FIG. 6. (The frequencies shown in this figure are nominal values only since the measured frequencies varied over a significant range due to non-linearities caused by bearing clearances.) In this figure the natural frequencies of the shafts are plotted against propeller rotational speed and are indicated by horizontal lines. Also shown are the shaft frequency line, the blade frequency line and the limits of the normal engine operating speed range. Where the shaft and blade frequency lines intersect the natural frequency lines potential resonant vibrations may be created. From FIG. 6 it may be seen that both the steel and bronze shafts have potential resonance frequencies in the operating range.

It may be concluded therefore that, as long as the input force remains unchanged, i.e. the propeller tip clearance is unaltered, potential vibration problems would still exist in the

operating range with either shaft. Since the only practical alternatives were to increase the diameter of the bronze shaft or to use a steel shaft it was concluded that it was not feasible to reduce the vibration in this way.

Even though initial vibration tests indicated that the steel shaft would not overcome the vibration problem, sea trials were still carried out with this shaft. As was expected, these results confirmed that the existing vibration levels were largely unchanged.

3.3 Reduction of bearing clearance or change of materials.

Initial tests were made with the bearing clearances reduced from 0.51 mm (0.02 in.) to 0.25 mm (.01 in.). Vibration measurements were taken on the 'P'-bracket bearing housing during operation, and these were analysed on a real-time spectrum analyser. The results are presented in FIGS. 2 & 5.

In Figure 2, traces A or B should be compared with trace C. It is clear that reducing the bearing clearance significantly reduces the overall vibration energy in the frequency range investigated. Subsequent tests indicated that the rate of bearing wear was also reduced, but was still not acceptable.

The tests on the steel shaft, however, had shown that, although the vibration levels had not altered appreciably, the bearing wear rate had been significantly reduced. Simultaneous tests conducted on a bronze shaft with sprayed Monel metal bearings also indicated a greatly reduced wear rate.

Since the vibration levels were largely unchanged for either of these tests, and the bearing clearances were still of the order of 0.51 mm (0.02 in.), it was concluded that the harder bearing materials (i.e. stainless steel or Monel metal) were more compatible with the Tufnol bearing and less prone to wear. Subsequent prolonged operating tests confirmed that both materials were satisfactory and had a low wear rate. The alternative adopted was to spray the bearing surfaces of the existing bronze shafts with monel metal.

It is important to stress, however, that the vibration level on the shafting is still high and the higher order whirl mode is still being excited. This mode is not expected to be troublesome in service.

4. PROPELLER RUDDER HYDRODYNAMIC INTERFERENCE.

Some questions were raised by the designers of the craft as to whether the vibration was caused entirely by propeller-hull interaction or also by propeller-rudder interaction.

Doubts were expressed about the overall propeller-hull-rudder clearances and it was considered worthwhile to carry-out some further tests to assess the criteria used in selecting these clearances.

During the tests vertical vibration measurements were made on the hull above the rudder posts and above the P-bracket, on both port and starboard sides, for the following operating conditions:

1. Port rudder removed and both engines operating through the speed range 1500-2200 RPM (See FIG. 7).

2. Both rudders allowed to trail so that they took up the local stream direction, both engines operating through the speed range 1500-2200 RPM (FIG. 8).

3. Both rudders set to zero degrees incidence, both engines operating through the speed range 1500-2200 RPM (FIG. 9).

The data obtained were analysed on a spectrum analyser as before and the responses obtained at blade frequency only are plotted against engine speed. This was considered to be an acceptable measure of blade-hull or blade-rudder interaction. As can be seen in figures 7-9 there is no indication that changing the rudder settings, or even removing a rudder, significantly changes the vibration levels at the measurement points. It is concluded therefore that the vibration is mainly caused by blade-hull interference and that the propeller-rudder clearances are acceptable.

5. CONCLUSIONS.

1. The severe vibration on the propeller shafts is caused by excitation of a high order whirling mode by propeller blade-hull interaction.

2. It was not practicable to reduce the vibration level by changing the shafting diameter, material or location.

3. In the presence of strong vibration Aluminium bronze and Tufnol bearing materials have a very poor performance.

4. Stainless steel or Monel metal and Tufnol are good materials for resisting a high level of bearing vibration and have a long operating life.

5. Reducing the bearing clearances significantly reduces the overall vibration level.

6. The propeller-rudder interference forces are not significant.

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1. Toms, A.E. and Martyn, D.K. — Whirling of line shafting Trans. I. Mar. E. Vol. 84, 1972. P. 176.

2. Ellis, R. — Stiffening of a boat propeller shaft bearing support bracket by a CFRP overlay. ARL SM NOTE 425 1976.

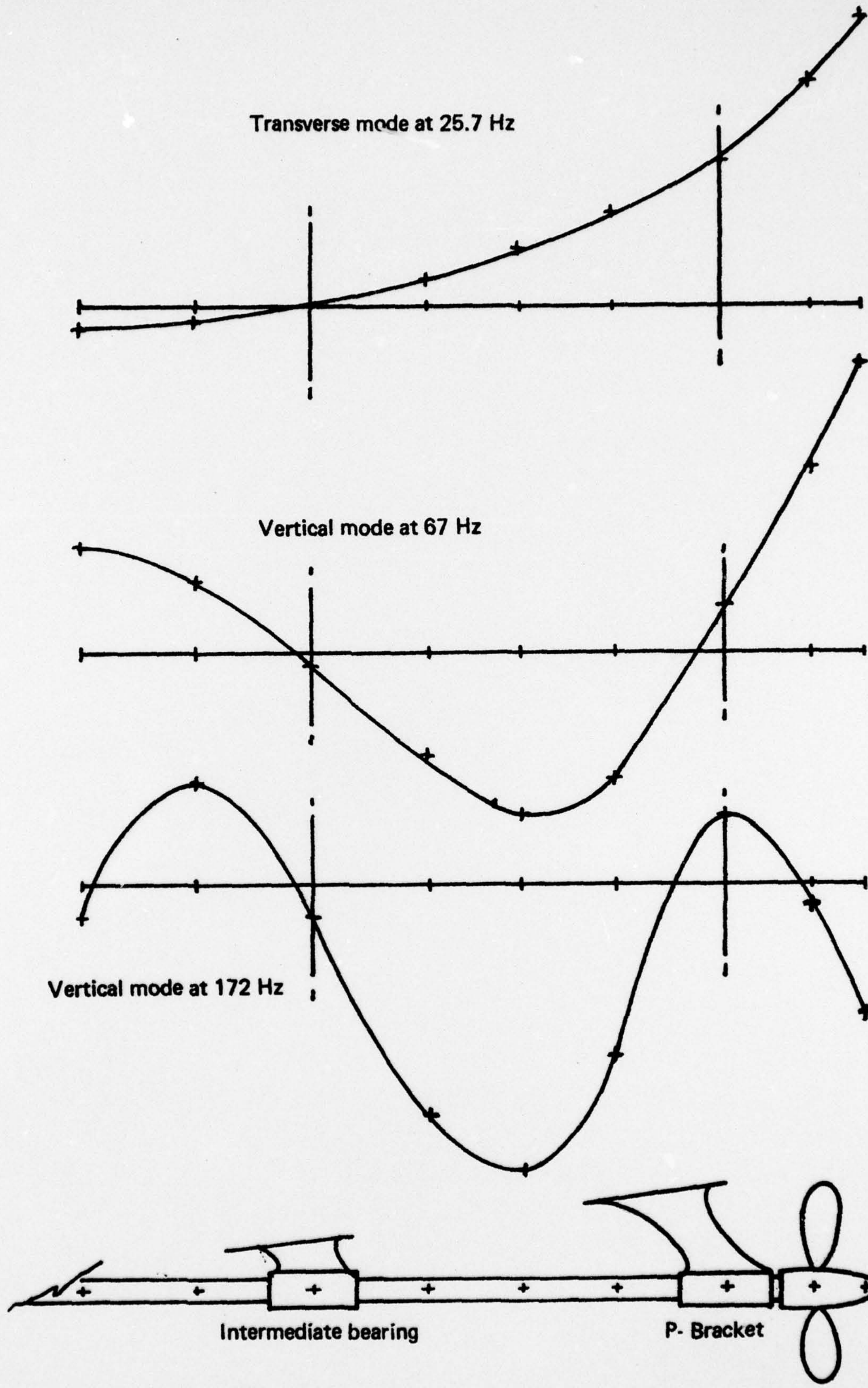

FIG. 1 MEASURED VIBRATION MODES OF ORIGINAL BRACKET

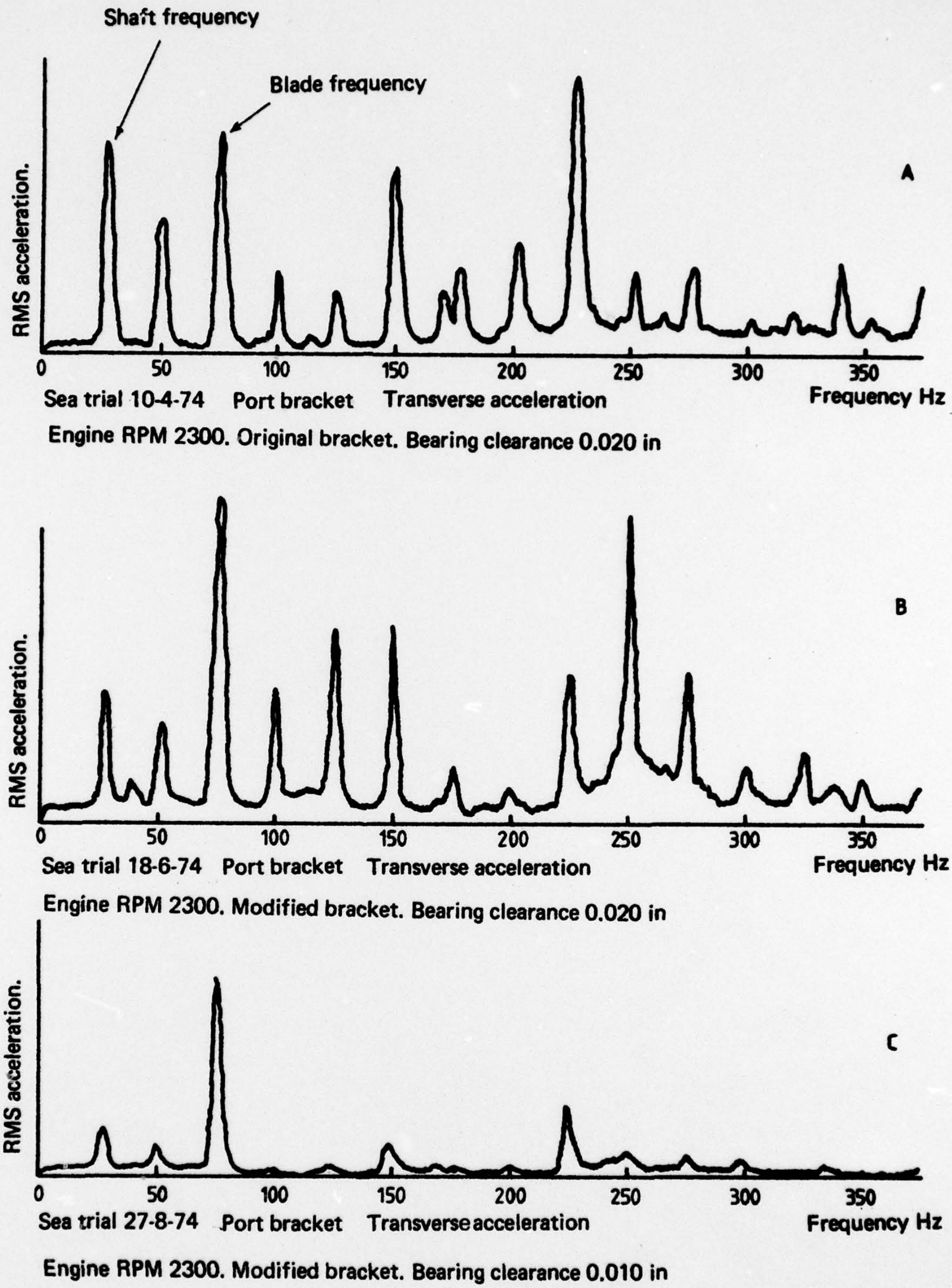

FIG. 2 FREQUENCY SPECTRA OF TRANSVERSE VIBRATION ON P- BRACKETS AT 2300 RPM

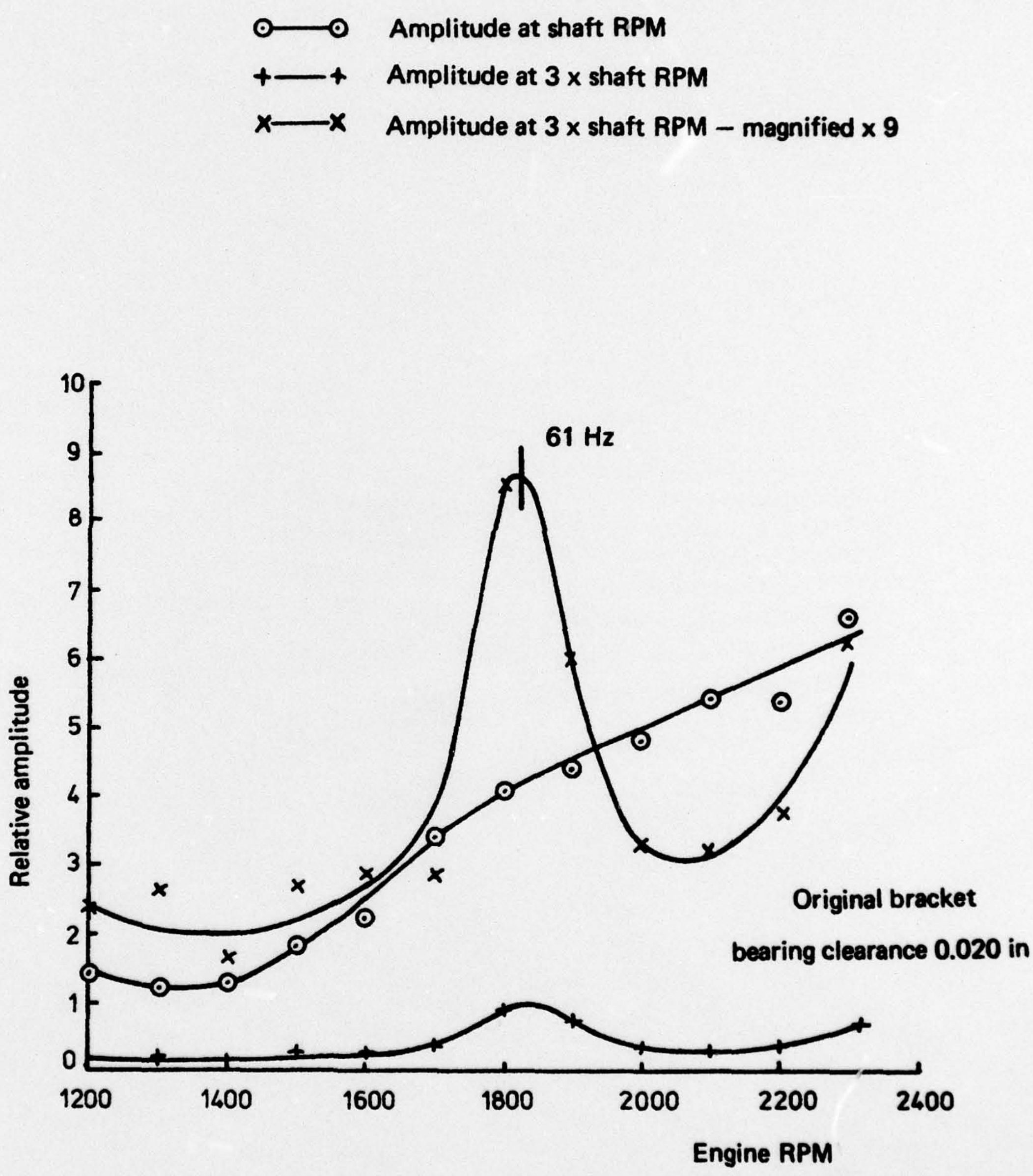

FIG. 3 VARIATION OF VIBRATION AMPLITUDE OF P- BRACKET WITH ENGINE RPM

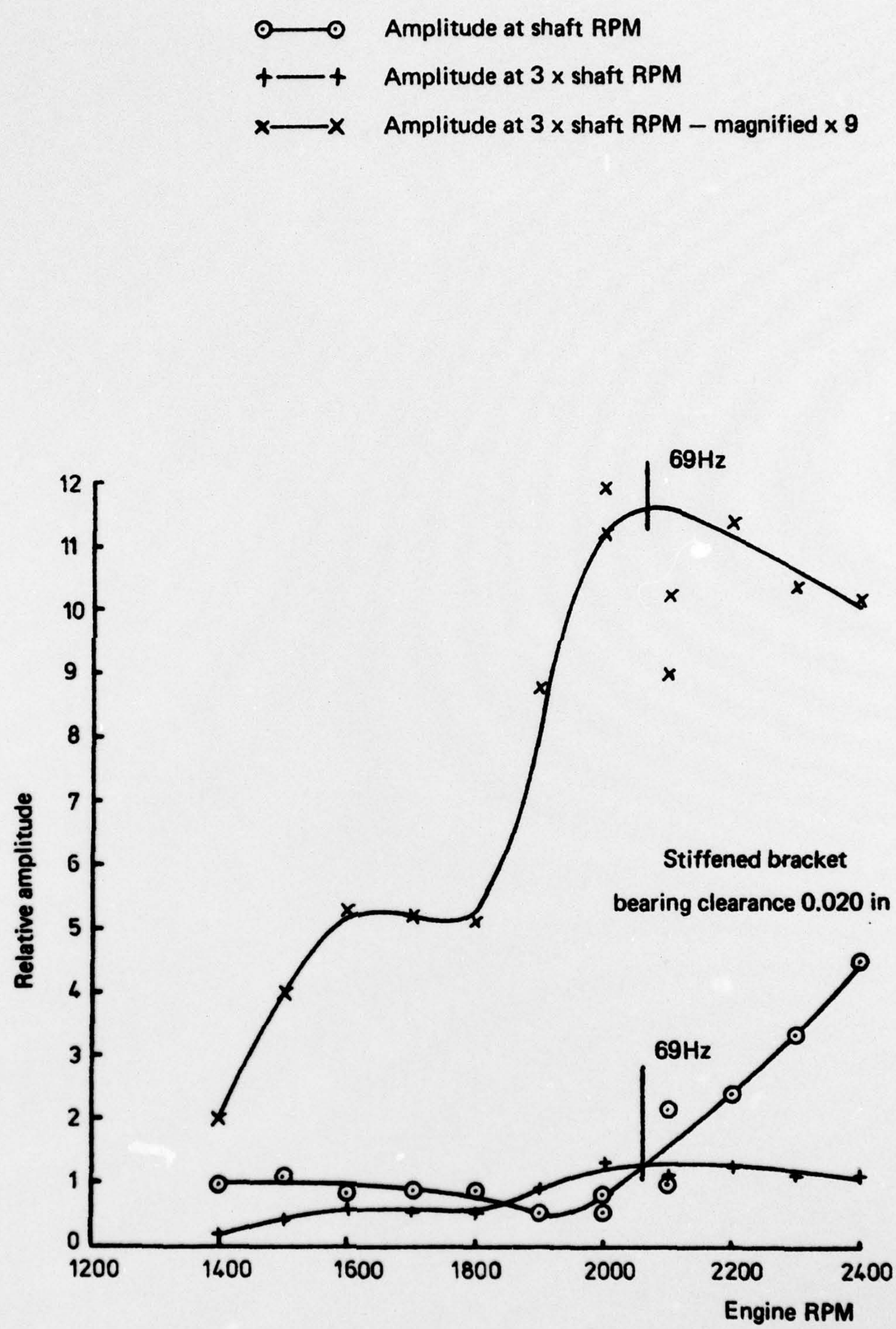

FIG. 4 VARIATION OF VIBRATION AMPLITUDE OF P- BRACKET WITH ENGINE RPM

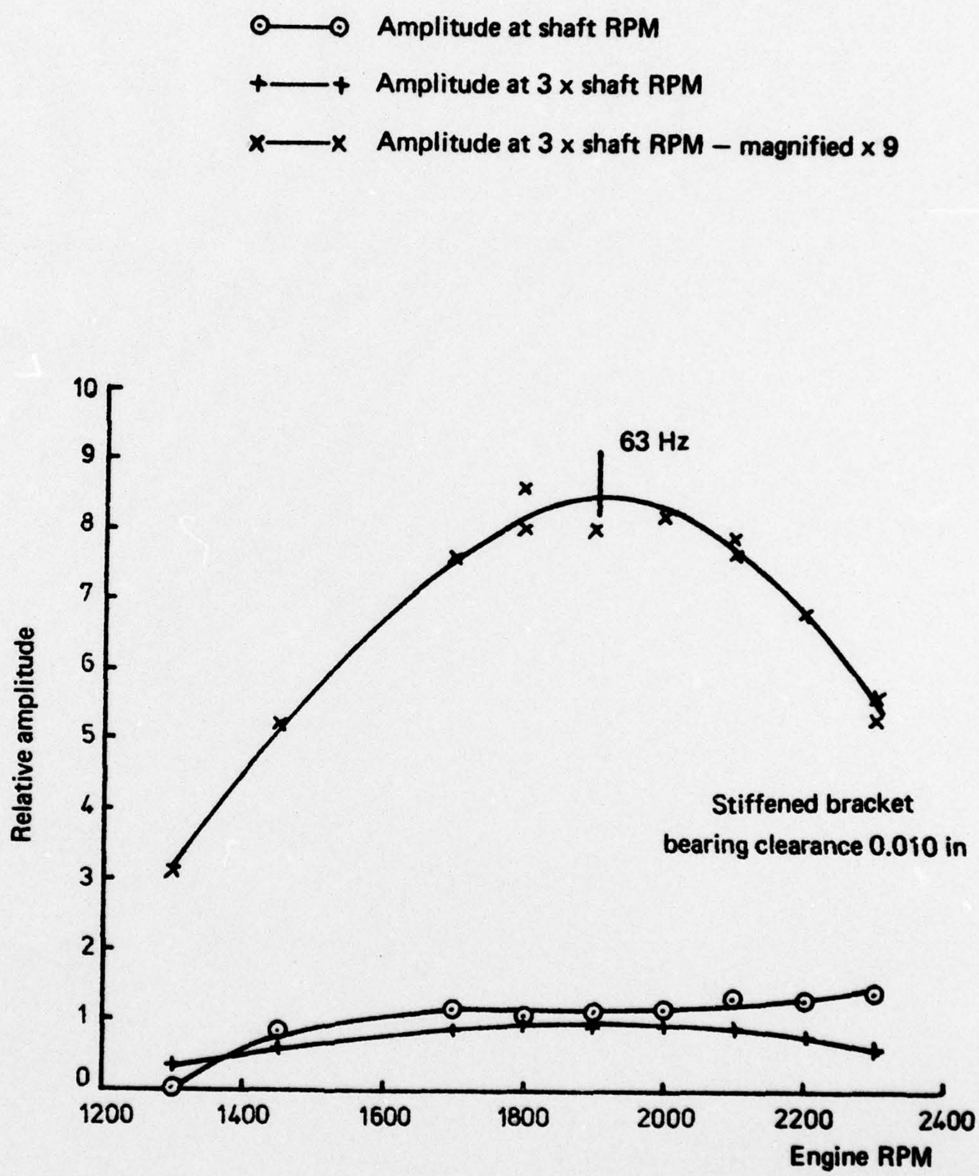

FIG. 5 VARIATION OF VIBRATION AMPLITUDE OF P- BRACKET WITH ENGINE RPM

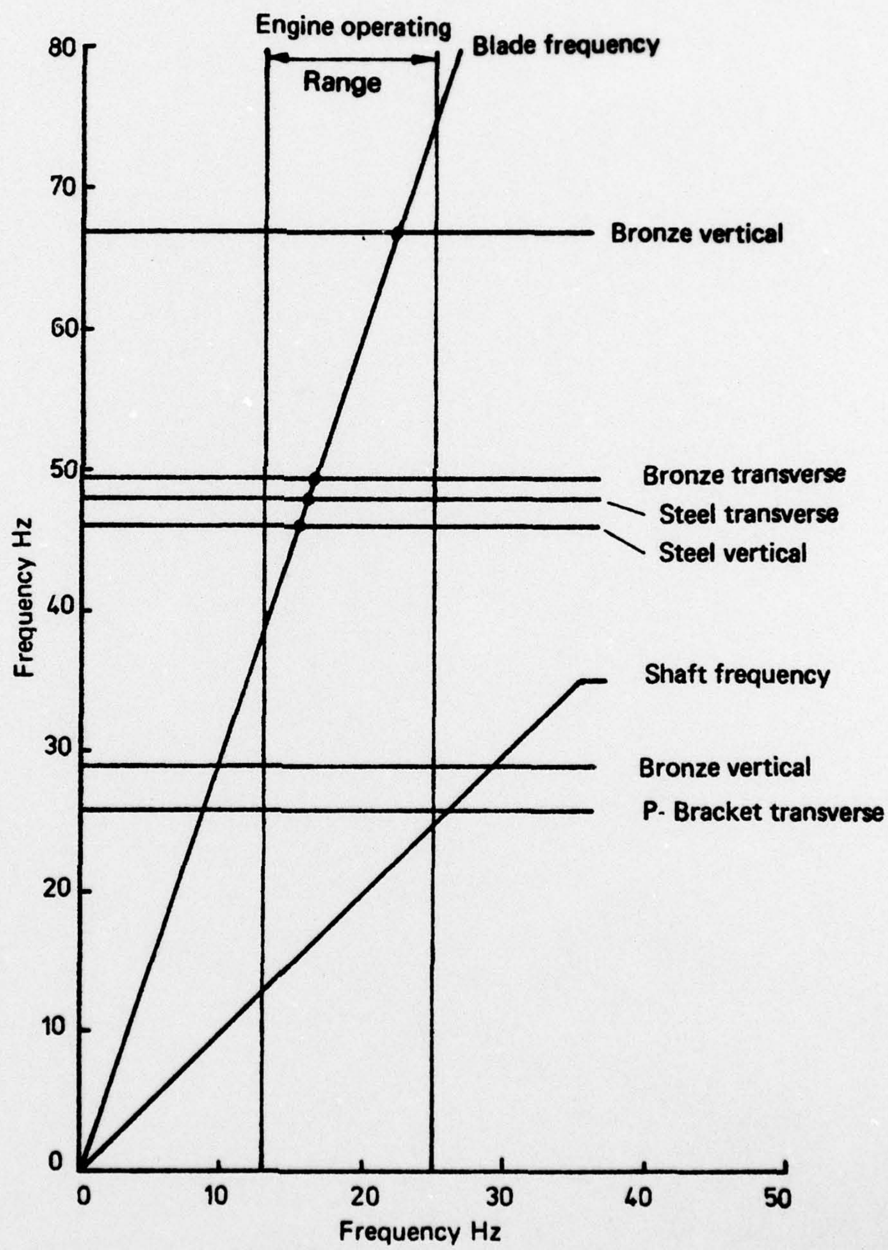

FIG. 6 COMPARISON OF NATURAL FREQUENCIES OF BRONZE AND STEEL SHAFTS

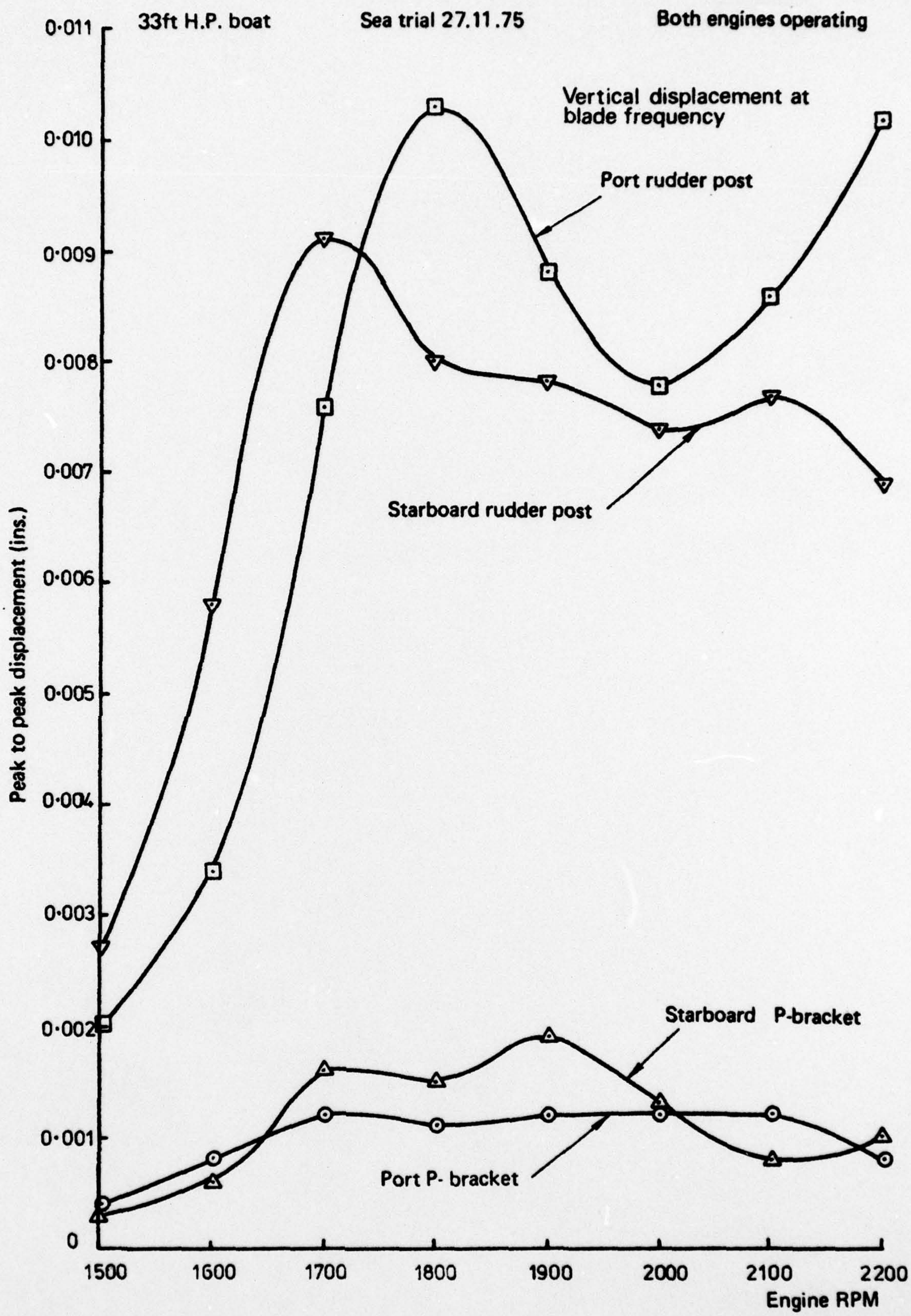

FIG. 7 COMPARISON OF HULL VERTICAL VIBRATION DATA
(Port rudder removed)

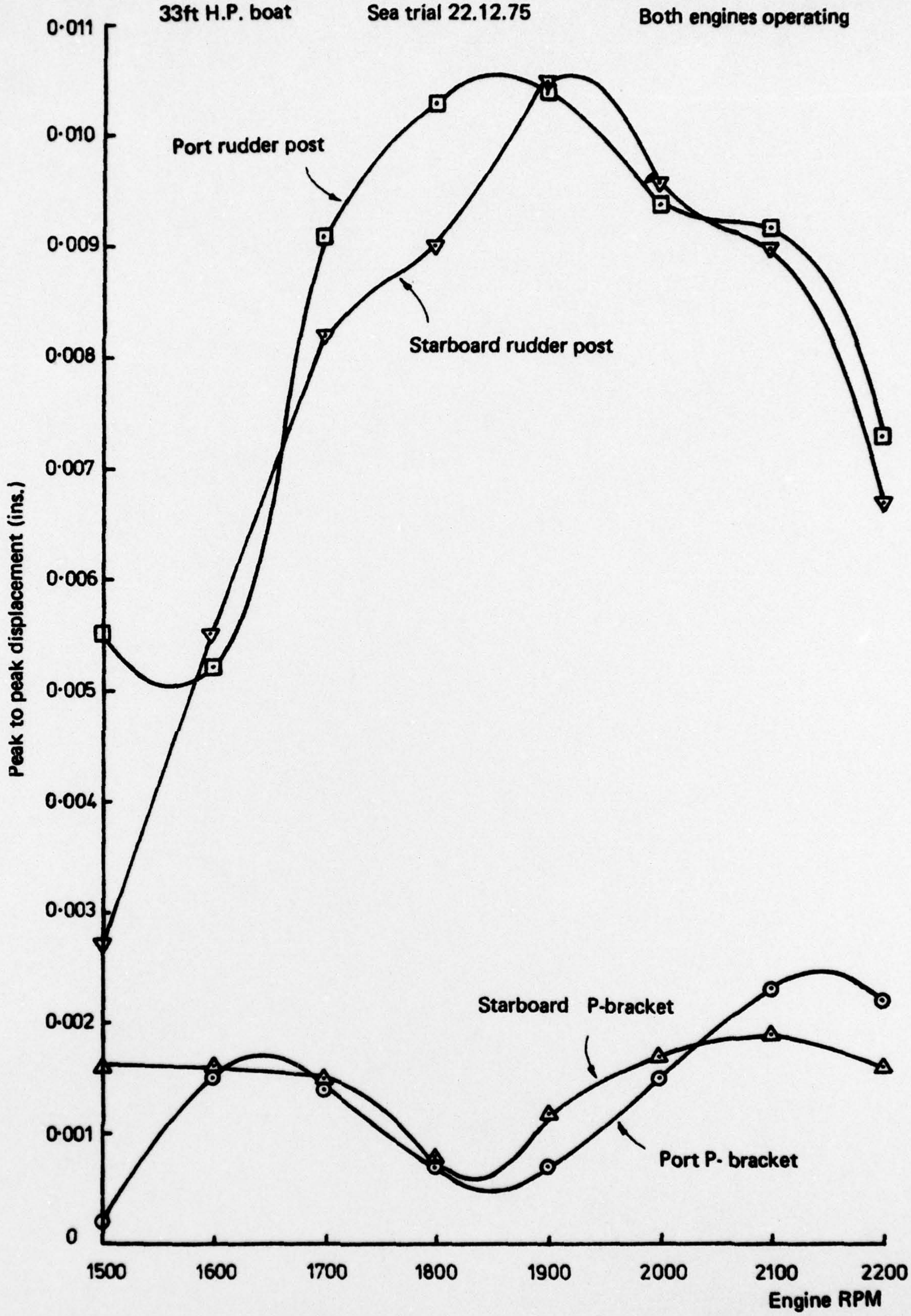

FIG. 8 COMPARISON OF HULL VERTICAL VIBRATION DATA
(Both rudders trailing)

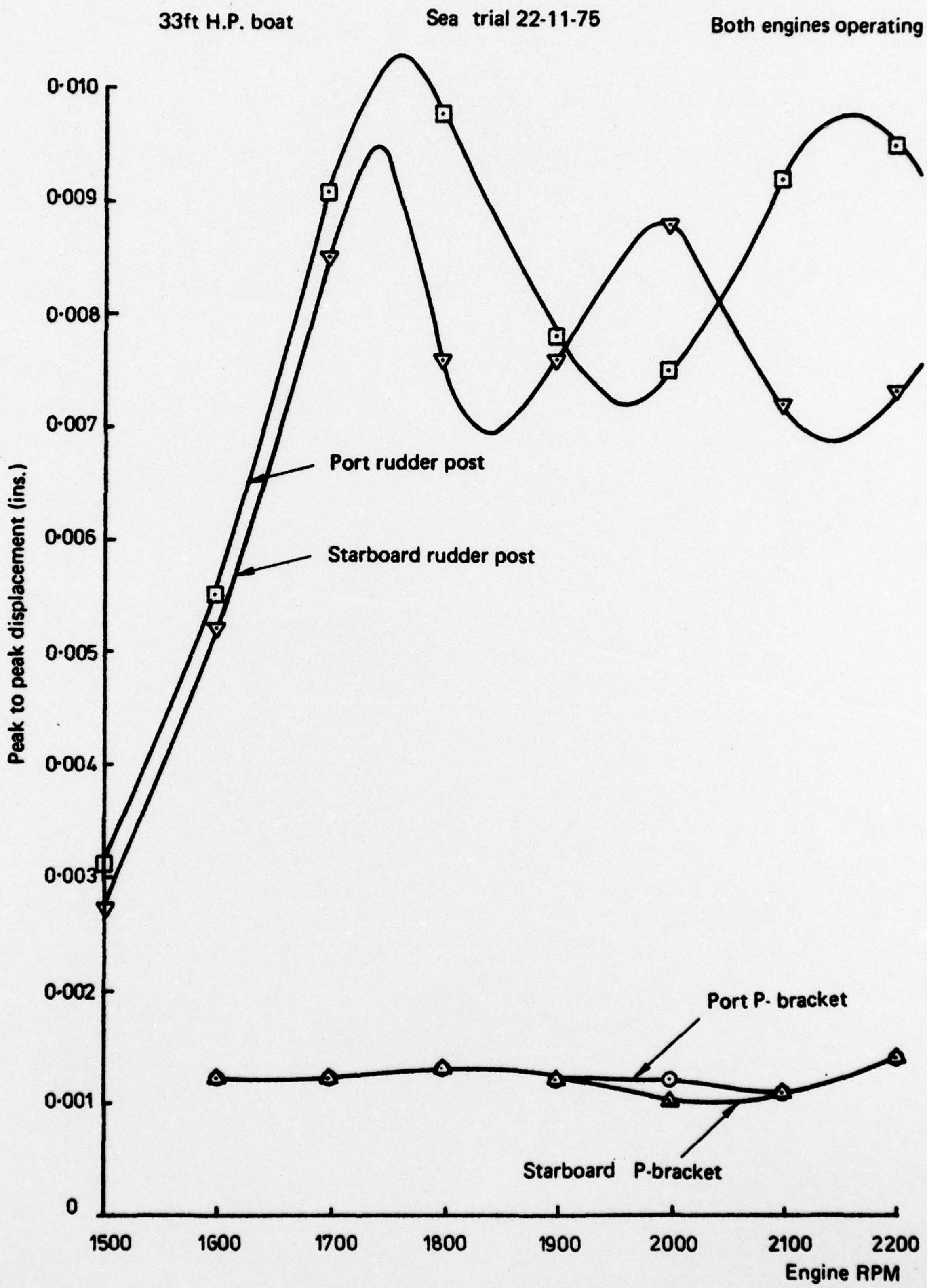

FIG. 9 COMPARISON OF HULL VERTICAL VIBRATION DATA
(Both rudders at zero incidence)

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